View metadata, citation and similar papers at core.ac.uk
brought to you by CORE
provided by K-State Research Exchange

Joshua Dotson, Ryan Cater, Tian Bai, Taylor Stein, Jason Steuber

Heat Sink Design Project

Spring 2014

Kansas State University

Heat Transfer

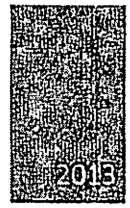
ME 573

Dr. Amy Betz

Abstract:

A single phase, microchannel heat sink and an experiment to test the heat sink is designed. The heat sink is designed to cool a 30mm x 10 mm plate that is subjected to a constant heat flux of 500 W/cm^2, a maximum surface temperature of 80 *C, and a maximum pressure drop of 300 kPa. The final design uses 47 micrometer x 292 micrometer silicone channels. Finally, the design uses water at a mass flow rate of 41.657*10^-3 kg/s and a temperature rise of 8.6*C.

Keywords: microchannel, heat sink, mechanical engineering, microfluidics



90

Heat Sink Design Project

ME 573

JOSH DOTSON, RYAN CATER, TIAN BAI, TAYLOR STEIN, JASON STEUBER

- lack of detail in experiments
- incorrect error calc.

W

Contents

Member Contributions

Ryan Cater – 20%

Developed equations for heat sink geometry and heat transfer. Ran all calculations analytical analysis. Developed graphs for Nussult number, average surface temperature and pressure drop. Found range of acceptable mass flow rates.

Jason Steuber – 20%

Wrote the section on Experimental design and wrote assumptions. Compiled all information into final document.

Josh Dotson – 20%

Wrote the problem statement. Wrote section on background information and motivation. Also found sources and developed in text citations.

Tian Bai – 20%

Developed equations to verify the constant heat flux boundary condition, find the heat loss to the environment as a function of temperature, the surface temperature, and the average Nu along the x direction.

Taylor Stein - 20%

Researched heat sink materials and working fluids. Provided help and research for other group members.

Project Description

Problem Statement

Our project was to design a single phase, microchannel heat sink and develop an experiment to test our model. The heat sink needs to cool a plate that is 30 mm x 10 mm that is subjected to a constant heat flux of 500 $\frac{W}{cm^2}$, a maximum surface temperature of 80 °C, and a maximum pressure drop of 300 kPa. We will choose the fin geometry and size, heat sink material, working fluid, and flow rate.

Using our model, we will create three plots. Each of the following values will be plotted against the mass flow rate of our working fluid.

1. Theoretical Nusselt number (Nu)
2. Average surface temperature
3. Pressure drop

Our experiment must specify where and how we will take measurements, as well as what types of probes we will use. Along with our experiment, we must develop equations to:

1. Verify the constant heat flux boundary condition
2. Find the heat loss to the environment as a function of temperature
3. The surface temperature
4. The average Nu along the x direction
5. The error on Nu using your designated temperature measurements.

We must verify that our Nu error is less than 10%.

Background and Motivation

Tuckerman and Pease pioneers in the field of microfluidics in the early 1980's. Modern electronics, which are more powerful and generate large amounts of heat, necessitated the development of more effective methods of cooling. Microchannels were developed to fulfill these needs. A cooling fluid runs through hundreds of microchannels, dissipating heat through forced convection. The small channels cause a decrease in the thickness of the thermal boundary layer, which generates a decrease in the convective resistance to heat transfer, thus generating high cooling rates [2].

Assumptions

(1) We can make θ_{cond} small by mounting the heat sink very close to the heat source.

(2) We can assume laminar flow.

(3) Flow is fully developed.

(4) We can conservatively assume that Nu has the minimum asymptotic value Nu_{∞}.[4]

(5) We will neglect heat transfer at the top and bottom of the channels.

(6) We will assume a constant heat transfer coefficient h up the walls which will allow us to get an analytical approximation for η. [4]

(7) The temperature will not vary across the width of the heat sink.

(8) We will neglect entry effects

Heat Sink Design

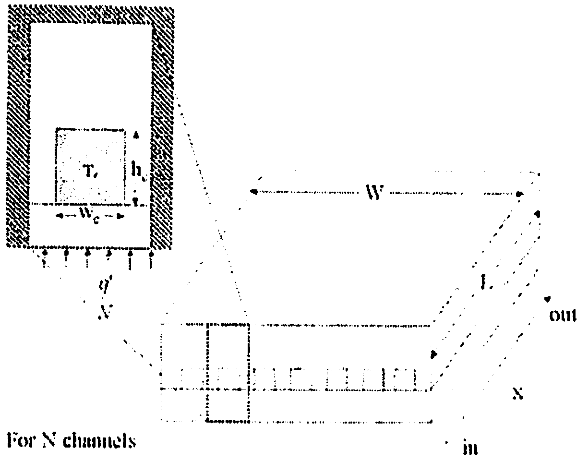


Figure 1: Heat Sink Design

Problem Constraints

$\Delta P := 300\ kPa$

$q'' := 500\ \frac{W}{cm^2}$

$L := 10\ mm$

$W := 30\ mm$

$T_{Smax} := 80\ ^\circ C$

Material Data

Material properties evaluated for lightly-doped Si at T=300K
Fluid properties evaluated for water at T=320K

$k_w := 148.1\ \frac{W}{m \cdot \Delta ^\circ C}$

$k_f := 640 \cdot 10^{-3}\ \frac{W}{m \cdot \Delta ^\circ C}$ $\quad \mu := 577 \cdot 10^{-6}\ \frac{N \cdot s}{m^2}$

$C_p := 4180\ \frac{J}{kg \cdot \Delta ^\circ C}$ $\quad \rho := \frac{1}{1.011 \cdot 10^{-3}}\ \frac{kg}{m^3}$

Optimum Dimensions

$Nu_{inf} := 6$ — Nusselt number across entire channel

$w_c := 2.29 \cdot \sqrt[4]{\frac{\mu \cdot k_f \cdot L^2 \cdot Nu_{inf}}{\rho \cdot C_p \cdot \Delta P}} = 47.079\ \mu m$ — Width of channel

$w_w := w_c$ — Width of fin

$\alpha := \sqrt[2]{\frac{k_w}{k_f \cdot Nu_{inf}}} = 6.21$ — Aspect Ratio

$h_c := \frac{\alpha \cdot (w_c + w_w)}{2} = 292.373\ \mu m$ — Height of channels

Experiment Design

Our heat sink will be clamped on to the heated surface and we will be sealing it with an O-ring. Between the clamp and the heated surface there will be Teflon for insulation. The heat sink and heated surface will be insulated with melamine foam to keep escaping heat to a minimum. We will be pumping the water through with a peristaltic pump. The liquid mass velocity can be obtained by measuring the volume of water exiting over an allotted time period. From here we can multiply the liquid mass velocity by the density of the liquid to give us mass flow rate. Inlet and outlet temperatures will be recorded using thermocouples (Type K, 0.5 mm 33 diameter, Omega, 100ms response time, ± 0.5°C uncertainty). We will use a pressure transducer (Honeywell, 15 psi/105.53 kPa, ± 0.087 psi / 0.61 kPa uncertainty, 100 μs response time) to measure the pressure drop along the channels. [1] The thermocouples will be placed directly in front and behind the length of the channels of the heat sink. We will be clamping a Pyrex plate on top of the heat sink that will give us an adiabatic tip condition. [4] Our design can be seen in figure 2.

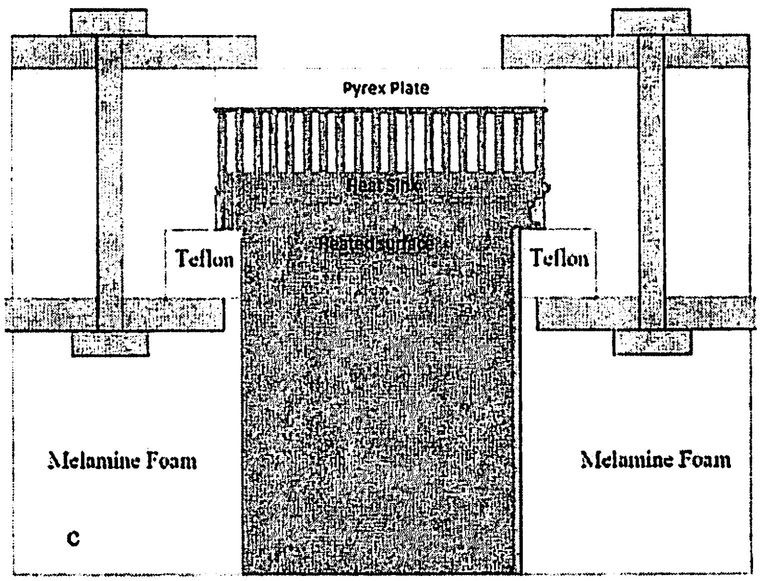
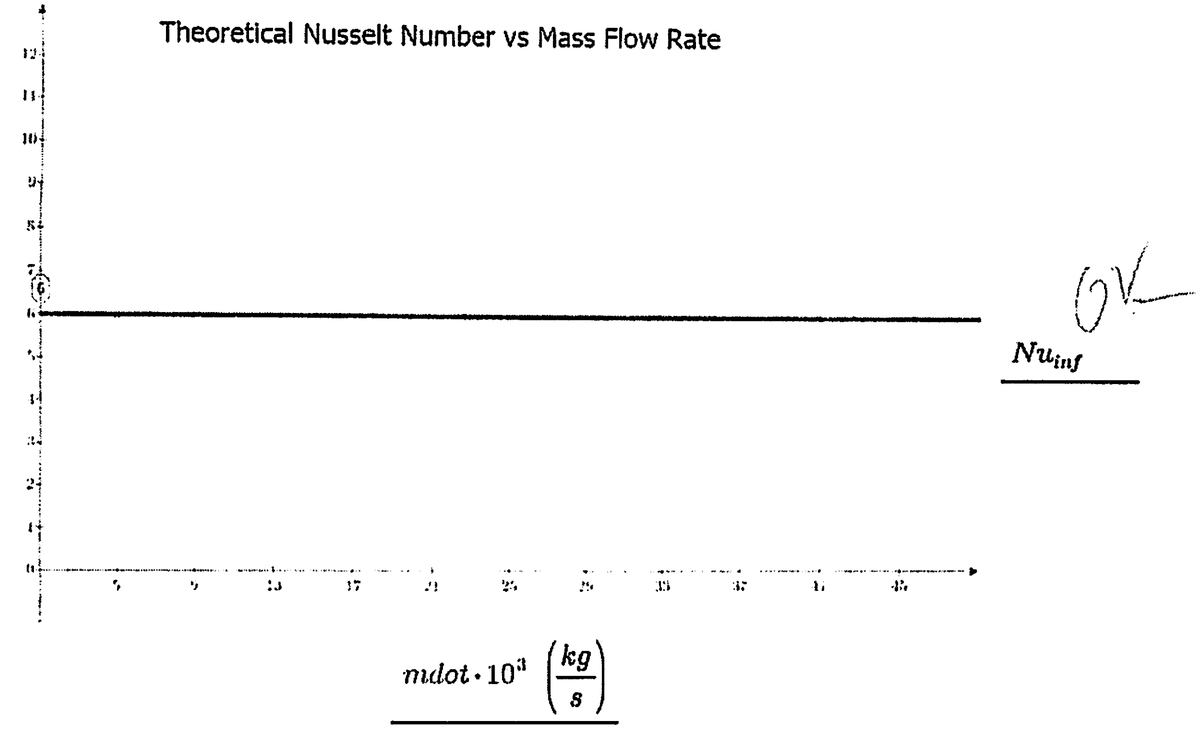


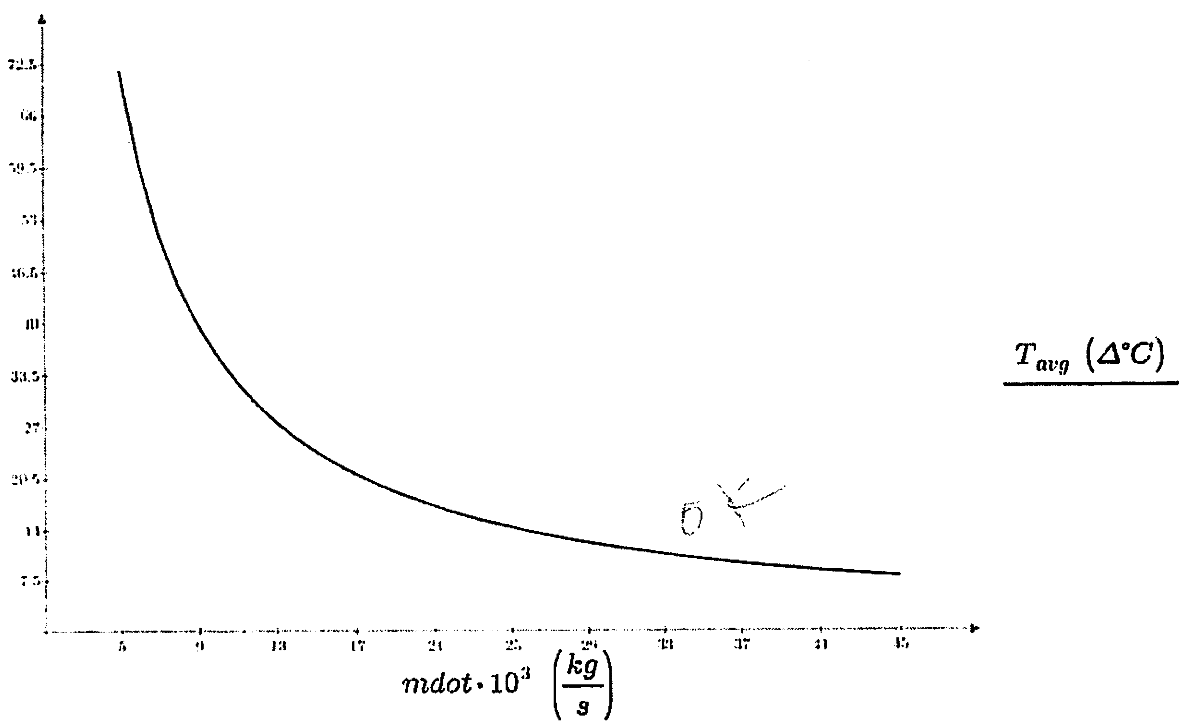
Figure 2: Cross- Sectional View of Test Section Adapted from [1] ok

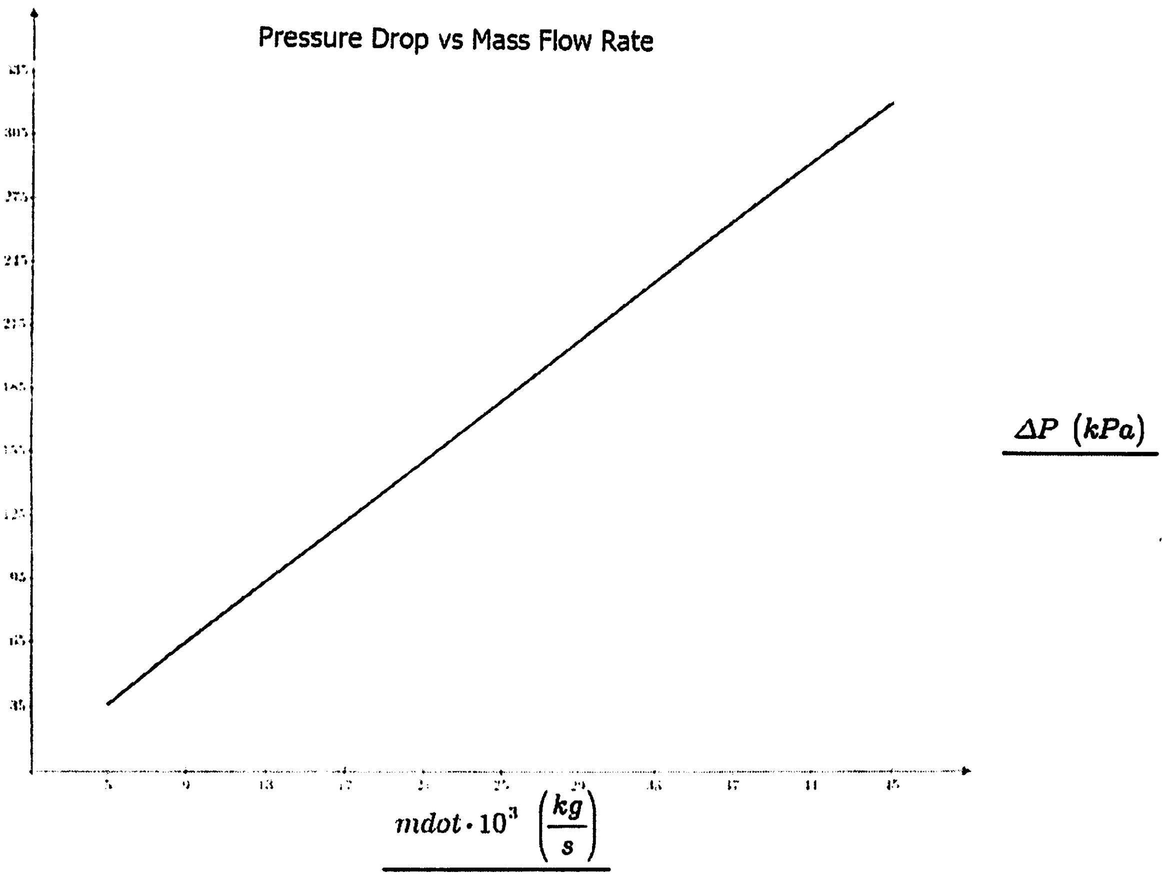
Results

Theoretical Nusselt Number vs Mass Flow Rate



Average Temperature vs Mass Flow Rate





Our range of possible mass flow rates to meet the specified operational constraints

$$mdot_{min} := \left(\frac{q'' \cdot L \cdot W}{C_p \cdot (T_{Sout} - T_{Lin}) - \theta_{conv} \cdot q'' \cdot W \cdot L} \right) = 6.636 \ \frac{kg \cdot (10^{-3})}{s}$$

$$mdot_{max} := \frac{1}{2} \cdot \rho \cdot v \cdot W \cdot w_c \cdot \alpha = 41.657 \ \frac{kg \cdot 10^{-3}}{s}$$

Our thermocouples give us an uncertainty of +-.5 degrees °C. Our fluid temperature difference at our optimum mass flow rate is 8.614 degrees C. This will give us an uncertainty in our Nusselt number of 5.8%.

No the error will be much higher.

Error propagation

Equations

Verify the constant heat flux boundary condition

OK

$$E_{loss} := \boxed{E} \cdot I - mdot \cdot C_p \cdot \Delta T_f$$

Where E and I are the voltage and current, respectively, from the heated surface's heating element.

Find the heat loss to the environment as a function of temperature

Surface temperature is found using the following equation.

$$q'' := h \cdot \left(\boxed{T_{surface}} - T_{fluid} \right)$$

The surface temperature

For constant q" and fully developed laminar flow, Nu is a constant for different cross sections.

The average Nu along the x direction

$$q'' := \frac{mdot \cdot C_p \cdot \boxed{\Delta T_{fluid}}}{A_c}$$

Summary/Discussion

We noticed that our Nusselt number is constant for our specified geometry regardless of the mass flow rate as long as the flow is fully developed and laminar. Increasing the mass flow rate lowers the average surface temperature and increases the pressure drop. The lower limit of mass flow rate is determined by the freezing temperature of the water inlet. The upper limit of the mass flow rate is determined by our maximum allowable pressure drop. As we increase mass flow rate our pressure drop also increases linearly.

References

[1] Betz, A. R., & Attinger, D. (2010). Can segmented flow enhance heat transfer in microchannel heat sinks? *International Journal of Heat and Mass Transfer, 53*(19-20), 3683-3691. Retrieved from www.scopus.com

[2] Hassan, I., Phutthavong, P., & Abdelgawad, M. (2004). Microchannel heat sinks: An overview of the state-of-the-art. Microscale Thermophysical Engineering, 8(3), 183-205. Retrieved from www.scopus.com

[3] Incropera, Frank P., Adrienne S. Lavine, David P. DeWitt, and Theodore L. Bergman. Fundamentals of Heat and Mass Transfer. John Wiley & Sons, 2011. Print.

[4] Tuckerman, D. B., & Pease, R. F. W. (1981). HIGH-PERFORMANCE HEAT SINKING FOR VLSI. Electron Device Letters, EDL-2(5), 126-129. Retrieved from www.scopus.com

Appendix I

Problem Constraints **Material Data**

$\Delta P := 300\ kPa$

$q'' := 500\ \frac{W}{cm^2}$

$L := 10\ mm$

$W := 30\ mm$

$T_{Smax} := 80\ °C$

Material properties evaluated for lightly-doped Si at T=300K
Fluid properties evaluated for water at T=320K

$k_w := 148.1\ \frac{W}{m \cdot \Delta °C}$

$k_f := 640 \cdot 10^{-3}\ \frac{W}{m \cdot \Delta °C}$ $\mu := 577 \cdot 10^{-6}\ \frac{N \cdot s}{m^2}$

$C_p := 4180\ \frac{J}{kg \cdot \Delta °C}$ $\rho := \frac{1}{1.011 \cdot 10^{-3}}\ \frac{kg}{m^3}$

Optimum Dimensions

$Nu_{inf} := 6$ Nusselt number across entire channel

$w_c := 2.29 \cdot \sqrt[4]{\frac{\mu \cdot k_f \cdot L^2 \cdot Nu_{inf}}{\rho \cdot C_p \cdot \Delta P}} = 47.079\ \mu m$ Width of channel

$w_w := w_c$ Width of fin

$\alpha := \sqrt[2]{\frac{k_w}{k_f \cdot Nu_{inf}}} = 6.21$ Aspect Ratio

$h_c := \frac{\alpha \cdot (w_c + w_w)}{2} = 292.373\ \mu m$ Height of channels

Optimized Heat Transfer and Max Possible Mass Flow Calculations

$D_h := 2 \cdot w_c = 94.158\ \mu m$

$h_{conv} := \frac{Nu_{inf} \cdot k_f}{D_h}$

$N := \sqrt[2]{\frac{2 \cdot h_{conv}}{k_w \cdot w_w}} \cdot h_c$

$\eta := \frac{\tanh(N)}{N} = 76.159\%$

$v := \frac{w_c^2 \cdot \Delta P}{12 \cdot \mu \cdot L}$

$$mdot_{max} := \frac{1}{2} \cdot \rho \cdot v \cdot W \cdot w_c \cdot \alpha = 41.657 \ \frac{kg \cdot 10^{-3}}{s}$$

$$\theta_{conv} := \frac{2 \cdot w_c}{k_f \cdot Nu_{inf} \cdot L \cdot W \cdot \alpha \cdot \eta}$$

$$\theta_{heat} := \frac{1}{mdot_{max} \cdot C_p}$$

$$\theta_{total} := \theta_{conv} + \theta_{heat} = 0.023 \ \frac{\Delta{}^{\circ}C}{W}$$

$$T_{Sout} := T_{Smax}$$

$$T_{Lin} := T_{Sout} - q'' \cdot W \cdot L \cdot \theta_{total}$$

$$T_{Lout} := T_{Lin} + q'' \cdot W \cdot L \cdot \theta_{heat}$$

$$T_{Sin} := T_{Lin} + q'' \cdot W \cdot L \cdot \theta_{conv}$$

$$\Delta T_S := T_{Sout} - T_{Sin}$$

$$\Delta T_L := T_{Lout} - T_{Lin}$$

	Surface	Liquid
Inlet	$T_{Sin} = 71.386 \ {}^{\circ}C$	$T_{Lin} = 45.464 \ {}^{\circ}C$
Outlet	$T_{Sout} = 80 \ {}^{\circ}C$	$T_{Lout} = 54.079 \ {}^{\circ}C$
Differential	$\Delta T_S = 8.614 \ \Delta{}^{\circ}C$	$\Delta T_L = 8.614 \ \Delta{}^{\circ}C$

Minimum Possible Mass Flow Calculations

$mdot_{min}$ realized when $T_{Lin} = 0\,{}^{\circ}C$ for water

$$T_{Lin} := 0 \ {}^{\circ}C$$

$$mdot_{min} := \left(\frac{q'' \cdot L \cdot W}{C_p \cdot \left(T_{Sout} - T_{Lin} - \theta_{conv} \cdot q'' \cdot W \cdot L \right)} \right) = 6.636 \ \frac{kg \cdot \left(10^{-3}\right)}{s}$$

$$\theta_{heat} := \frac{1}{mdot_{min} \cdot C_p}$$

$$\theta_{total} := \theta_{conv} + \theta_{heat}$$

$$T_{Sout} := T_{Smax}$$

$$T_{Lin} := T_{Sout} - q'' \cdot W \cdot L \cdot \theta_{total}$$

$T_{Lout} := T_{Lin} + q'' \cdot W \cdot L \cdot \theta_{heat}$

$T_{Sin} := T_{Lin} + q'' \cdot W \cdot L \cdot \theta_{conv}$

$\Delta T_S := T_{Sout} - T_{Sin}$

$\Delta T_L := T_{Lout} - T_{Lin}$

	Surface	Liquid
Inlet	$T_{Sin} = 25.921\ {}^{\circ}C$	$T_{Lin} = 0\ {}^{\circ}C$
Outlet	$T_{Sout} = 80\ {}^{\circ}C$	$T_{Lout} = 54.079\ {}^{\circ}C$
Differential	$\Delta T_S = 54.079\ \Delta{}^{\circ}C$	$\Delta T_L = 54.079\ \Delta{}^{\circ}C$

Graphics Calculations

$i := 5..45$

$$mdot_{i-5} := \frac{45 \cdot 10^{-3}}{45} \cdot i \cdot \frac{kg}{s}$$

$$T_{avg} := \frac{q'' \cdot W \cdot L}{mdot \cdot C_p}$$

$$\Delta P := \frac{mdot \cdot 24 \cdot \mu \cdot L}{\rho \cdot w_c^{\ 3} \cdot W \cdot \alpha}$$

3-0235 — 50 SHEETS — 5 SQUARES
3-0236 — 100 SHEETS — 5 SQUARES
3-0237 — 200 SHEETS — 5 SQUARES
3-0137 — 200 SHEETS — FILLER
COMET

$$\theta_{tot} = \frac{T_{sout}}{q''WL}$$

$$\theta_{heat} = \frac{T_{sout}}{q''WL} - \theta_{conv}$$

$$\frac{1}{\dot{m} c_p} = \frac{T_{sout}}{q''WL} - \theta_{conv}$$

$$\dot{m} = c_p\left(\frac{T_{sout}}{q''WL} - \theta_{conv}\right)$$

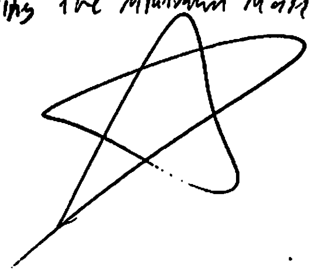
$$q''WL\,\theta_{tot} = T_{sout} - T_{Lin}$$

$$\theta_{heat} = \frac{T_{sout} - T_{Lin}}{q''WL} - \theta_{conv}$$

$$\frac{1}{\dot{m}} = c_p\left(\frac{T_{sout} - T_{Lin} - \theta_{conv}\, q''WL}{q''WL}\right)$$

$$\dot{m} = \frac{q''WL}{c_p\left(T_{sout} - T_{Lin} - \theta_{conv}\, q''WL\right)}$$

This page is the hand calcs for finding the minimum mass flowrate



3-0235 — 50 SHEETS — 5 SQUARES
3-0236 — 100 SHEETS — 5 SQUARES
3-0237 — 200 SHEETS — 5 SQUARES
3-0137 — 200 SHEETS — FILLER
COMET

$$T_{avg}(mdot) = T_{s,out} - T_{s,in} = T_{s,out} - \left[T_{s,out} - q''WL\left(\theta_{conv} + \frac{1}{mdot\, c_p}\right) + q''WL\theta_{conv}\right]$$

$$T_{s,in} = T_{L,in} + q''WL\theta_{conv} = T_{s,out} - q''WL\left(\theta_{conv} + \frac{1}{mdot\, c_p}\right) + q''WL\theta_{conv}$$

$$T_{L,in} = T_{s,out} - q''WL\,\theta_{tot} = T_{s,out} - q''WL\left(\theta_{conv} + \frac{1}{mdot\, c_p}\right)$$

$$\theta_{tot} = \theta_{conv} + \theta_{heat} = \theta_{conv} + \frac{1}{mdot\, c_p}$$

$$\theta_{heat} = \frac{1}{mdot\, c_p}$$

These calculations find $T_{avg}(mdot)$

$$T_{avg}(mdot) = q''WL\left(\cancel{\theta_{conv}} + \frac{1}{mdot\, c_p} - \cancel{\theta_{conv}}\right)$$

$$T_{avg}(mdot) = \frac{q''WL}{mdot\, c_p}$$

$$V = \frac{W_c^2 \Delta P}{12 \mu L}$$

These calculations find $\Delta P(mdot)$

$$mdot = \frac{1}{2}\rho V W w_c \alpha$$

$$mdot = \frac{1}{2}\rho \frac{W_c^2 \Delta P}{12 \mu L} W w_c \alpha$$

$$\Delta P = \frac{\cancel{2}\, mdot \cdot 24 \cdot \mu L}{\rho W_c^3 W \alpha}$$